

High-power Stirling-type pulse tube cooler working below 30 K

D.M. Sun^{a,b,c}, M. Dietrich^{a,b}, G. Thummes^{a,b,*}

^aInstitute of Applied Physics, University of Giessen, D-35392 Giessen, Germany

^bTransMIT-Centre for Adaptive Cryotechnology and Sensors, D-35392 Giessen, Germany

^cInstitute of Refrigeration and Cryogenics, Zhejiang University, Hangzhou 310027, China

ARTICLE INFO

Article history:

Received 7 May 2009

Accepted 19 June 2009

Keywords:

E. Pulse tube
E. Regenerators
E. Stirling-type cooler
F. Power applications

ABSTRACT

High-power Stirling-type pulse tube coolers (PTCs) are promising candidates for cooling HTS devices and gas liquefaction or separation applications. Nevertheless, till now most high-power Stirling-type PTCs are not able to reach a refrigeration temperature below 35 K. Here, a high-power two-stage Stirling-type PTC was designed, manufactured and experimentally investigated. In order to realize a convenient coupling with a thermal load, U-shape configuration is adopted in both stages, which makes it more challenging to distribute the gas flow and reduce dead volume in the cold end heat exchanger. By optimizing operating conditions, flow straightener, and double-inlet opening, the cooler has reached no-load refrigeration temperatures of 29.6 K and 27.1 K at 55 Hz and 40 Hz, respectively. Furthermore, the cooler is able to provide cooling powers of 50 W at 45.6 K and 100 W at 59.3 K when input *pV* powers are 4.77 kW and 4.59 kW, respectively.

© 2009 Elsevier Ltd. All rights reserved.

1. Introduction

For future commercial applications of high-temperature superconducting (HTS) devices, such as transformers, motors, generators, cables, fault current limiters, and magnetic energy storage systems, there is urgent need for efficient cryocoolers with low maintenance and high reliability [1]. Among the different types of closed-cycle cryocoolers, high-frequency (40–60 Hz) Stirling-type PTCs with oil-free linear compressors are considered as the preferred choice for HTS power applications, since this kind of cooler has a high potential to fulfill the above requirements.

With the availability of large-scale linear compressors with electrical power consumption in the range of 4–20 kW, the development of high-power Stirling-type PTCs has started a few years ago [2–9]. Up to now only single-stage systems with inline configuration of pulse tube and regenerator have been realized. The inline configuration avoids losses from curved gas flow and dead volume at the junction between pulse tube and regenerator. However, in inline configuration the cold heat exchanger is located in the middle of the cold head, which makes thermal interfacing rather difficult.

Most of the previous work on high-power Stirling-type PTCs aimed for refrigeration temperatures in the range of 55–80 K [2–5,8,9] which is appropriate for HTS power applications in rather low magnetic fields. Recently, Praxair Inc. has developed a high-power PTC for future cooling of HTS transmission cables that

provides a cooling power of 1 kW at 77 K by use of a 20 kW linear compressor [9].

For high magnetic field applications, like in HTS-motors, -generators and -magnets, lower operation temperatures of 25–30 K are required [1,6]. This is a consequence of the field dependence of the critical current of the presently available HTS BSCCO tapes that form the superconducting windings in the machine. The reduction of the critical current of these BSCCO tapes in a magnetic field is much less pronounced at 20–30 K than at 77 K and therefore much less HTS conductor is needed in a machine that operates below 30 K (e.g. [6]).

In a previous ministry funded project, we have developed a single-stage inline PTC operated on a 10 kW linear compressor [6,7] for potential cooling of a 4 MVA HTS-machine made by Siemens AG [10]. The challenging design target of that cooler was a cooling power of 80 W at 30 K. However, first tests of that cooler revealed a strong azimuthal temperature inhomogeneity around the circumference of the regenerator indicating significant losses from re-circulating streaming in the stainless steel mesh regenerator with large cross-sectional area. The onset of the regenerator streaming, which limited the minimum temperature to above 50 K, was found to depend on the temperature gradient along the regenerator as well as the magnitude of mass flow [6]. By increasing the transverse heat conduction in the wire mesh regenerator, the streaming losses were considerably reduced and a minimum temperature of 34.6 K was reached. In that modified version, the cooler provided cooling powers of 50 W at 45 K and 100 W at 56 K with an electric input power of 6.3 kW [7].

The above single-stage PTC could not meet the requirements for cooling of HTS-machines at 30 K or below. In order to reach lower

* Corresponding author. Address: Institute of Applied Physics, University of Giessen, D-35392 Giessen, Germany. Tel.: +49 641 99 33460; fax: +49 641 99 33409.

E-mail address: thummes@transmit.de (G. Thummes).

refrigeration temperatures, we have been developing a two-stage Stirling-type PTC, which is the subject of the present paper. This new cooler development is part of a government supported joint project coordinated by Siemens AG. In contrast to all of the aforementioned high-power Stirling-type PTCs, this new cooler exhibits a U-shaped configuration of the regenerator and pulse tube, since this arrangement facilitates the thermal coupling to the HTS-machine. By optimizing the flow straighteners at the cold end heat exchangers, the opening of the double-inlet valve at the second stage and the operating conditions of the system, a no-load refrigeration temperature of 27.1 K has been obtained. Main computational and experimental results are presented in this paper.

2. Experimental setup and measurement system

The PTC refrigeration system consists of a commercial linear compressor with dual opposed pistons (QDrive model 42SM-2S297 W) and an in-house made two-stage thermally coupled pulse tube cold head. The compressor has a nominal electric input power of 10 kW. The main dimensions of the cold head, such as the length and diameter of the pulse tube, regenerator, inertance tube, and transfer line, were designed by the commercial modelling software Sage [11]. Since the diameter of the second stage regenerator is rather large (about 77 mm), the one-dimensional Sage program is not able to simulate flow distribution problems. To solve this shortcoming, three-dimensional CFD simulation by use of Fluent 6.2 was used as assistant tool to analyze the flow conditions and to design the cold heat exchangers.

As schematically shown in Fig. 1, the refrigeration system is characterized by inter-stage thermal coupling via a copper bridge, U-shape configuration and a long transfer line (length: 1 m) between the cold head and compressor. Each stage of the cold head consists of aftercooler, regenerator, cold end heat exchanger, pulse tube, warm end heat exchanger, and phase shifter. Fig. 2 shows the photograph of the cold head. Unlike usual designs, the geometric

dimensions of the first stage are smaller than those of the second stage, so as to focus pV power into the second stage. The diameter and length of the first stage regenerator are 29 mm and 52 mm, respectively. The diameter and total length of the second stage regenerator are 77 mm and 79 mm, respectively. Both phase shifters are composed of inertance tubes and a buffer volume. Two sets of phase shifters, named PS_A and PS_B, were tested. PS_A is optimized for operation at about 55 Hz, which is close to the optimum working frequency of the compressor. PS_B, optimized for 45 Hz, is used to test the performance of the cold head at lower frequency. In the following, phase shifter PS_A is used unless otherwise stated.

In order to control DC flow and to efficiently adjust the temperature profiles along the regenerator and pulse tube of the second stage, two anti-parallel needle valves are adopted as double-inlet for the second stage. The aftercoolers and warm end heat exchangers are water cooled.

Flow distribution and heat transfer are of equal importance in the cold end heat exchanger of a high-power Stirling-type PTC, especially in case of U-shaped configuration. Several kinds of flow straighteners have been tested, including slot-type copper blocks, stacks of diffusion bonded copper screens, and their combinations. For the cold head under consideration, a conic stack of diffusion bonded copper screens turned out to be the best choice.

The compressor is equipped with two position sensors (X_1 and X_2) and a pressure sensor to measure the generated acoustic power, also referred to as pV power W_{pV} . Additional pressure sensors are installed at the entrance of the regenerators and close to the warm end heat exchanger of the second stage. For evaluation of the efficiency of compressor and cold head, the electrical input power W_e is also monitored. In order to obtain real-time values for phases and amplitudes, the pressure, position and electrical wave form data are measured by two synchronized A/D cards. The refrigeration temperature T_{C2} is measured at the cold end heat exchanger of the second stage by means of a calibrated Cernox resistance thermometer. The refrigeration temperature T_{C1} of the

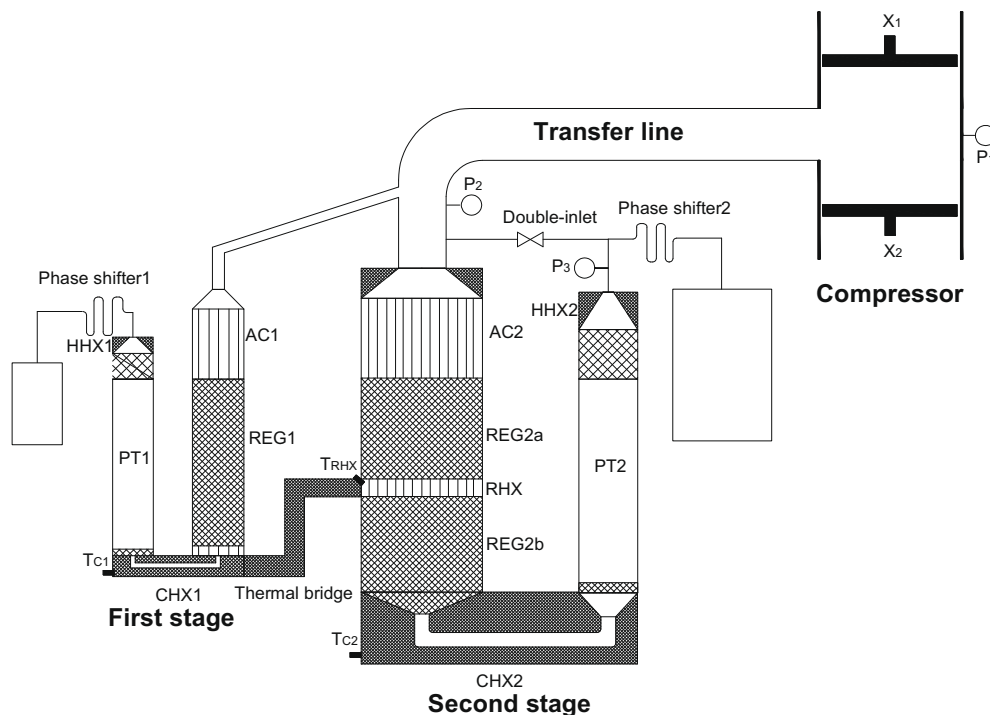


Fig. 1. Schematic of the two-stage Stirling-type PTC with linear compressor. AC1, AC2: first and second stage aftercooler; REG1: first stage regenerator; REG2a, REG2b: two parts of the second stage regenerator; RHX: heat exchanger in the middle of the second stage regenerator; CHX1, CHX2: first and second stage cold end heat exchanger; PT1, PT2: first and second stage pulse tube; HHX1, HHX2: first and second stage hot end heat exchanger; P_1 , P_2 , P_3 : pressure sensors; T_{C1} , T_{C2} , T_{RHX} : temperature sensors.

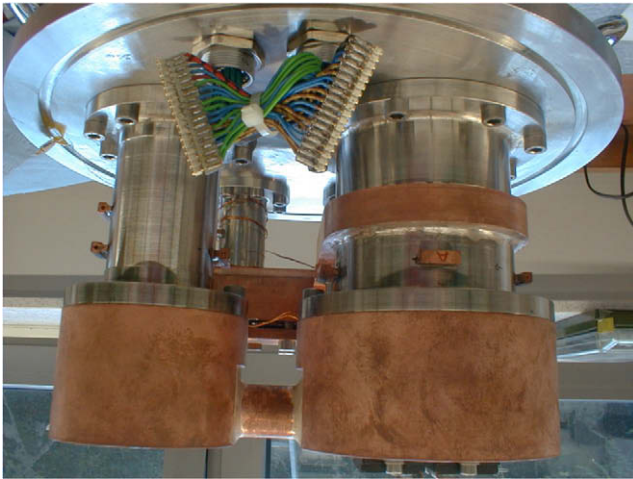


Fig. 2. Photograph of the PTC cold head.

first stage is measured by a Pt-100 resistance thermometer. Additionally, a number of Pt-100 sensors are installed along the regenerator and pulse tube wall of the second stage to measure the temperature profiles.

3. Experimental results and discussion

3.1. Pressure amplitude amplifying effect of transfer line

As shown in Fig. 1, the PTC cold head is connected to linear compressor by a 1 m long transfer line. According to thermoacoustic theory, the transfer line is able to amplify the pressure amplitude when its length is less than one fourth of the acoustic wavelength, while consuming some of the power flow [12,13]. From the point of view of acoustic impedance, it functions as a matching component between compressor and cold head. Fig. 3 shows the pressure amplitude amplifying effect of the transfer line by experiments and simulation. The simulation, based on Sage, is performed throughout the whole system composed of cold head and compressor. As seen from Fig. 3, at all input pV powers the transfer line increases the pressure amplitude by 0.25–0.3 bar. According to the simulation, the loss inside the transfer line is less than 4% of the transported acoustic power. The simulation results are in good agreement with the experiments in amplitude amplification (i.e. pressure amplitude difference). The offset of the computed pressure curves towards higher pressures by about 0.3 bar is ascribed to some acoustic impedance difference of the cold head between in simulation and experiment. As seen from the lower part of Fig. 3, the transfer line changes the phase of pressure oscillation by about 4° . The phase shift from simulation is less than 1° higher than the experimental one, and both of them are nearly independent of pV power.

3.2. Effect of pre-cooling temperature

As shown in Fig. 1, the two stages are coupled by a thermal bridge made from copper. The ends of the thermal bridge are tightly screwed to the two heat exchangers. Indium foil is sandwiched between the contact surfaces to further reduce the thermal resistance. Fig. 4 shows the effect of input pV power on the temperature (T_{C1}) of the cold end heat exchanger of the first stage and on the temperature (T_{RHx}) of the middle heat exchanger of the second stage. It is seen that the temperature difference between the two ends of the thermal bridge increases with input power, which

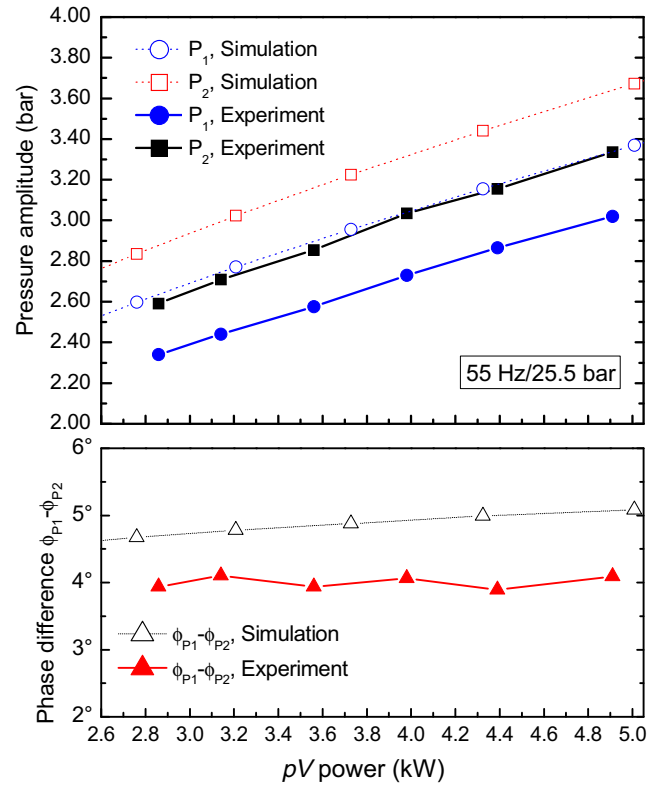


Fig. 3. Pressure amplitude and phase shift along the transfer line as function of pV power from experiment and simulation.

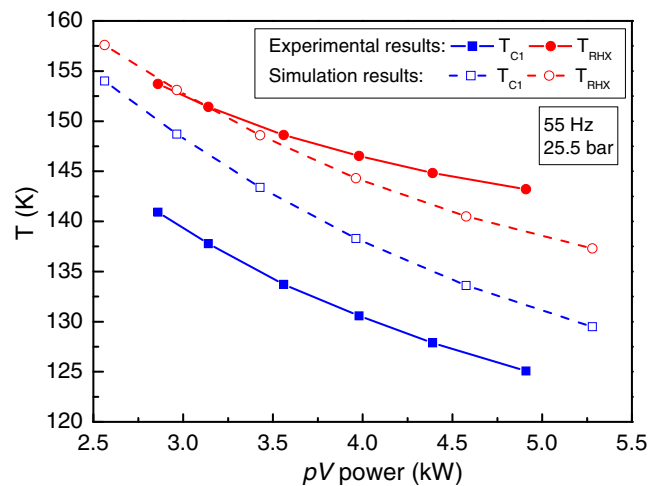


Fig. 4. Effect of input pV power on pre-cooling temperature.

indicates increased pre-cooling power. The temperature difference ($T_{RHx} - T_{C1}$) is about 12.8 K when the pV power W_{pV} is 2.86 kW, and it goes up to 18.1 K with W_{pV} of 4.9 kW. Taking into account the dimensions of the thermal bridge [14] and the thermal properties of copper, heat conduction calculation shows that the pre-cooling power through the thermal bridge is 31 W and 44 W with W_{pV} of 2.86 kW and 4.9 kW, respectively, which is in good agreement with the separate cooling capacity test of the first stage PTC. The simulation results in Fig. 4 show similar increase of the temperature difference with pV power. Since the loss in the second stage regenerator in simulation is less than that in experiment, the tem-

perature difference along the thermal bridge is considerably smaller in simulation.

3.3. Cool-down curves

Fig. 5 shows the cool down process of the two-stage PTC. In order to demonstrate the effect of input power on its performance, input power was increased gradually. Initially, the working frequency was set to 55 Hz. At a working pressure of about 21 bar, the linear compressor works resonantly and thus more efficiently at a lower frequency. Therefore, when the refrigeration temperature T_{C2} reached about 33 K the frequency was lowered from 55 Hz to 53 Hz. Then, after the refrigeration temperature reached 32.2 K, the double-inlet valves (shown as DIV in Fig. 5) were opened and optimized. As shown, the temperature of the cold end heat exchanger of the second stage reached 29.3 K and then stabilized at 29.6 K. At the same time, the cold end heat exchanger temperature of the first stage stabilized at 134.2 K. It should be noted that the double-inlet can only help to decrease the refrigeration temperature of the second stage PTC by about 2.9 K, which is not as remarkable as with small-scale Stirling-type PTCs, e.g. [15]. This is because the acoustic power and mass flow going into the inertance tube of a large-scale Stirling-type PTC are large enough to achieve the required phase shift. In the present PTC, the main benefit of the DIV comes from its effect on the temperature profile along the regenerator and pulse tube by introducing a small DC flow, as verified by the simulation.

3.4. Load curves

Fig. 6 shows the load curves of the PTC at 53 Hz, 21.2 bar and 3.0 kW pV power with and without DIV. The cooler reaches no-load refrigeration temperatures of 29.6 K and 32.0 K with and without DIV, respectively. The cooler provides 50 W of cooling power at about 50 K with and without DIV, and 120 W at 77.5 K without DIV. It is obvious that the double-inlet helps to increase the thermal efficiency of the cooler below 50 K. In addition, Fig. 6 shows the computed load curve without DIV, which exhibits a slope that is close to that of the experimental load curve. However, in simulation the no-load temperature is lower (27 K) and consequently the cooling powers are higher than the measured ones.

In order to increase the cooling capacity, the cooler was also operated at a higher filling pressure of 25 bar and a higher working frequency of 55 Hz without DIV. As shown in Fig. 7, at a medium pV power of 4.0 kW the cooler reached no-load temperature of 33 K.

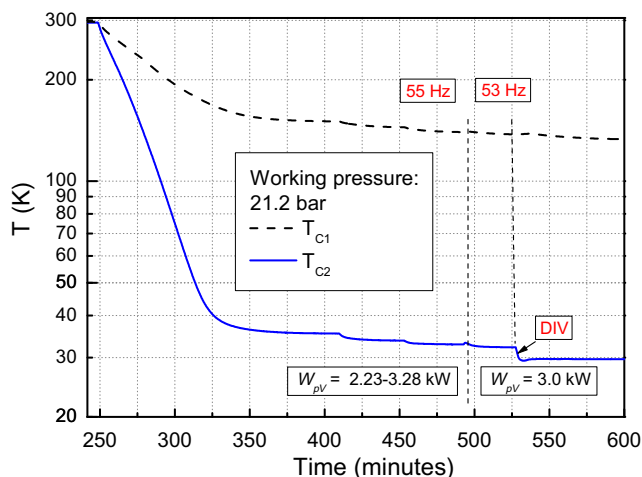


Fig. 5. Cool down curves of the two-stage PTC.

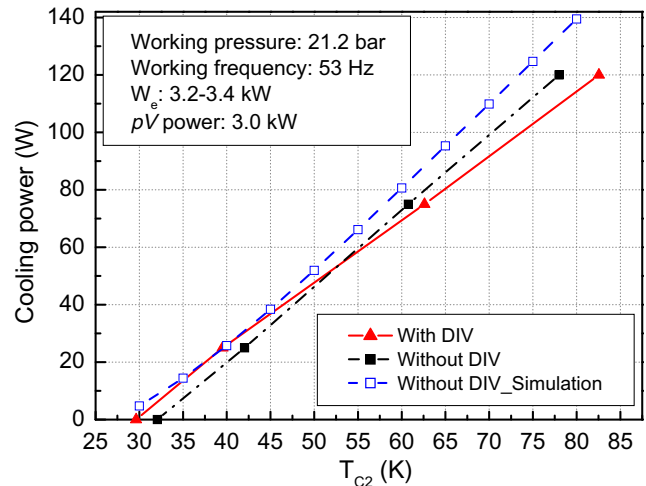


Fig. 6. Load curves at 53 Hz and 21.2 bar with and without double-inlet.

Maximum cooling capacity is achieved at full stroke of the compressor. Due to the limitation of piston stroke, in this test the cooler was operated at constant piston stroke of 90% of full stroke in order to obtain the largest cooling capacity at different refrigeration temperatures. It is seen from Fig. 7 that the cooler is able to supply cooling powers of 50 W and 100 W at 45.6 K and 59.3 K when the pV powers are 4.77 kW and 4.59 kW, respectively.

For comparison, Fig. 7 also displays the cooling power of the previously realized single-stage cooler [7]. It is seen that the present cooler exhibits a higher cooling performance below 45 K.

It is also seen from Fig. 7 that the simulation (without DIV) reflects the experimental load curves rather well above 45 K, while strong deviations occur at lower temperatures. The reason is that below 40 K the cooler shows apparent temperature inhomogeneities in the second stage regenerator, which strongly degrade the performance of the cooler [14]. Therefore the cooler cannot reach the computed no-load refrigeration temperature of about 25 K in Fig. 7.

3.5. Effect of working frequency

The performance of the two-stage PTC is strongly influenced by the working frequency. As shown in Fig. 8, it is much easier to get

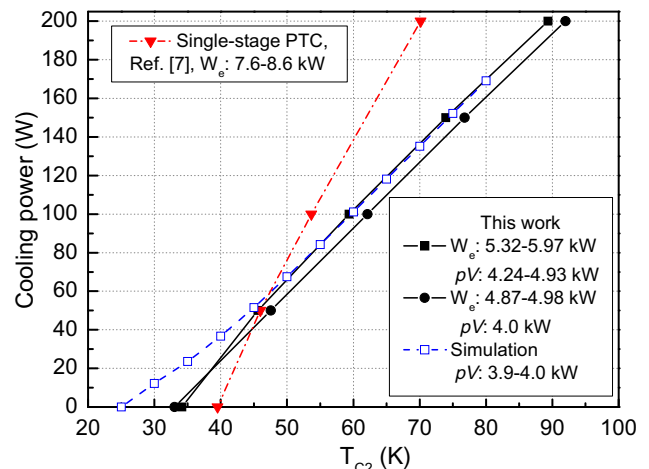


Fig. 7. Load curves at 55 Hz and 25 bar of the present two-stage PTC without double-inlet; a load curve of the single-stage PTC from Ref. [7] is shown for comparison.

lower refrigeration temperature at a lower working frequency, which is due to the improved thermal transfer conditions and lower regenerator loss. However, the no-load refrigeration temperature difference between 53 Hz and 55 Hz is much smaller than that between 55 Hz and 58 Hz, which indicates that for this cooler much lower refrigeration temperature cannot be obtained by only further reducing the frequency.

In order to reach lower refrigeration temperature the phase shifters that were optimized for about 55 Hz (inertance tubes: PS_A) should be modified for operation at a lower frequency. So, the cooler was operated at 40 Hz with a new set of inertance tubes (PS_B). It should be noted that the compressor is designed to work efficiently at 60 Hz and 25 bar average pressure. In order to make it transmit as much pV power as possible before reaching piston stroke and electrical current limit, the filling pressure was reduced to 15.2 bar. After these modifications, the two-stage PTC reached a no-load refrigeration temperature of 29.5 K without double-inlet valve (see Fig. 9). By optimizing the double-inlet, the cooler reached 26.2 K first and then stabilized at about 27.1 K, as seen from Fig. 9. According to the load curves, at a working frequency of 40 Hz the PTC is able to supply about 40 W of cooling power at 51 K with only 2 kW of pV power, which corresponds to 9.5% of Carnot efficiency.

The variation of refrigeration temperature with pV power in Fig. 8 exhibits a temperature minimum for 53 and 55 Hz working frequency. Most probably, such a minimum also exists for the 58 Hz curve at higher pV powers which are out of the accessible measurement range.

The rise of the minimum no-load temperature towards higher pV powers indicates the onset of parasitic losses caused by internal streaming. As already mentioned in Section 1, in Refs. [6,7], such a kind of loss caused by re-circulating streaming in the stainless steel mesh regenerator with large cross-sectional area was revealed, and could be related to the weak transverse thermal coupling in the regenerator matrix. Because the regenerator diameter of the present two-stage PTC is considerably smaller than that of the former single-stage PTC [6,7], and because additional brass screens were sandwiched with the stainless steel screens in the regenerator in order to increase the transverse thermal conduction, the problem with weak transverse heat transfer is not so evident in the present case.

By detailed circumferential temperature measurements around the middle heat exchanger (RHX in Fig. 1) and second stage regenerator (REG2b in Fig. 1), a temperature inhomogeneity caused by

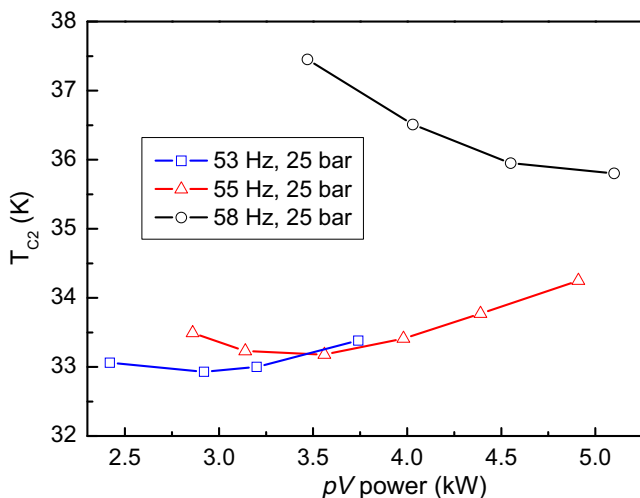


Fig. 8. Refrigeration temperature versus pV power at different working frequencies.

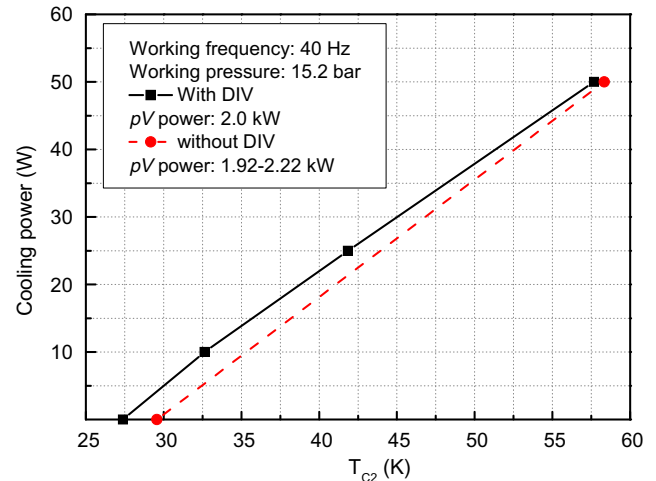


Fig. 9. Load curves at 40 Hz and reduced pressure of 15.2 bar with and without double-inlet.

asymmetric pre-cooling at the inter-stage thermal bridge was identified as another root of the observed streaming [14]. This internal streaming that grows with increasing pV power and decreasing refrigeration temperature; degrades the performance of the cooler. The onset of internal streaming is also considered as the reason why the PTC cooled to 26.2 K, but then stabilized at a higher no-load temperature of 27.1 K (Fig. 9).

4. Conclusions

The special features of the present high-power Stirling-type PTC are a two-stage design with thermal conduction coupling of the stages, U-shaped configuration of the pulse tubes and regenerators, and a rather long transfer line between compressor and cold head. Great efforts were needed to eliminate the unfavourable effects from curved gas flow in the U-shaped configuration.

A minimum no-load temperature of 27.1 K has been achieved, which is the first time for a high-power Stirling-type PTC to reach a temperature below 30 K.

While the present PTC reached a rather low temperature, it is supposed that the PTC design with small first and large second stage connected via a thermal bridge is not able to reach the required base temperature below 20 K. The large second stage does not only make the system susceptible to regenerator streaming, but also requires a large dead volume at the cold end to provide sufficient flow straightening in the U-shaped configuration.

Problems with regenerator streaming in a large cold head could be avoided by the use of a number of smaller PTC cold heads operating in parallel on the same compressor. Such an arrangement of parallel cold heads is expected to reach no-load temperatures below 20 K and to provide the required cooling power near 25 K.

Acknowledgements

This work is supported in part by the German Ministry of Economics and Technology (BMWi) under Grant no. 03SX221A. D.M. Sun thanks the German Academic Exchange Service (DAAD) for providing a scholarship.

References

- [1] Gromoll B. Technical and economical demands on 25–77 K refrigerators for future HTS-series products in power engineering. *Advances in cryogenic engineering*, vol. 49. New York: AIP; 2004. p. 1797–804.

- [2] Zia JH. A commercial pulse tube cryocooler with 200 W refrigeration at 80 K. *Cryocoolers*, vol. 13. New York: Springer; 2005. p. 165–71.
- [3] Yuan J, Maguire J. Development of a single stage pulse tube refrigerator with linear compressor. *Cryocoolers*, vol. 13. New York: Springer; 2005. p. 157–63.
- [4] Potratz SA, Nellis FG, Maddocks JR, Kashani A, Helvensteijn BPM, Rhoads GL, et al. Development of a large-capacity Stirling-type pulse tube refrigerator. *Advances in cryogenic engineering*, vol. 51. New York: AIP; 2006. p. 3–10.
- [5] Tanchon J, Ercolani E, Trolhier T, Ravex A, Poncet JM. Design of a very large pulse tube cryocooler for HTS cable applications. *Advances in cryogenic engineering*, vol. 51. New York: AIP; 2006. p. 661–8.
- [6] Gromoll B, Huber N, Dietrich M, Yang LW, Thummes G. Development of a 25 K pulse tube refrigerator for future HTS-series products in power engineering. *Advances in cryogenic engineering*, vol. 51. New York: AIP; 2006. p. 643–52.
- [7] Dietrich M, Yang LW, Thummes G. High-power Stirling-type pulse tube cryocooler: observation and reduction of regenerator temperature inhomogeneities. *Cryogenics* 2007;47:306–14.
- [8] Tanchon J, Trolhier T, Ravex A, Ercolani E. Prototyping a large capacity high frequency pulse tube cryocooler. *Cryocoolers*, vol. 14. Boulder: ICC Press; 2007. p. 133–9.
- [9] Potratz SA, Abbott TD, Johnson MC, Albaugh KB. Stirling-type pulse tube cryocooler with 1 kW of refrigeration at 77 K. *Advances in cryogenic engineering*, vol. 53. New York: AIP; 2008. p. 42–8.
- [10] Frank M et al. High-temperature superconducting rotating machines for ship applications. *IEEE Trans Appl Supercond* 2006;16:1465–8.
- [11] Gedeon D. Sage: object orientated software for stirling-type machine design. In: *Proceedings of the 29th Intersociety Energy Conversion and Engineering Conference*, vol. 4. Monterey CA: American Institute for Aeronautics and Astronautics; 1994. p. 1902–07.
- [12] Sun DM, Qiu LM, Wang B, Xiao Y. Transmission characteristics of acoustic amplifier in thermoacoustic engine. *Energy Convers Manage* 2008;49:913–8.
- [13] Hu JY, Dai W, Luo EC. Thermoacoustically driver pulse tube coolers with acoustic amplifiers. *Advances in cryogenic engineering*, vol. 51. New York: AIP; 2006. p. 1564–71.
- [14] Sun DM, Dietrich M, Thummes G, Qiu LM. Investigation on regenerator temperature inhomogeneity in stirling-type pulse tube cooler. *Chin Sci Bull* 2009;54(6):986–91.
- [15] Yang LW, Thummes G. Development of Stirling-type coaxial pulse tube cryocoolers. *Cryocoolers*, vol. 13. New York: Kluwer Academic/Plenum Publishers; 2005. p. 141–8.